Minimizing the levelized cost of electricity production from low-temperature geothermal heat sources with ORCs: water or air cooled?☆

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Abstract

A system optimization of ORCs cooled by air-cooled condensers or wet cooling towers and powered by low-temperature geothermal heat sources is performed in this paper. The configuration of the ORC is optimized together with the geometry of all the components. The objective is to minimize the levelized cost of electricity (LCOE) and the performance of ORCs with different types of cooling systems are compared to each other. The results show that it is economically more interesting to use mechanical-draft wet cooling towers instead of air-cooled condensers. The difference in performance is especially large for a low brine-inlet temperature. The investment cost of wet cooling towers is much lower than the one of air-cooled condensers, so the discount rate has less influence on the former type of cooling. The effect of the water price and the climate conditions on the economics of ORCs is also investigated. Both the brine-inlet temperature and the dry-bulb temperature of the surroundings have a strong influence and values of the optimized LCOE between about 55 and 185 €/MWh are obtained.

Keywords: ORC, geothermal, air-cooled condenser, wet cooling tower, economics, optimization

1. Introduction

It is expected that low-temperature geothermal heat sources will be used more often in the future for electricity production [1, 2]. One issue with these sources is that the conversion efficiency to electricity is low due to the low temperature of the source. Many researchers have tried to maximize this efficiency by optimizing the performance of organic Rankine cycles (ORCs), but the absolute efficiency remains low due to the Carnot limit. Most of the research on ORCs focuses on the optimization of the thermodynamic cycle. Simple cycles, recuperated cycles and cycles with turbine bleeding are proposed, they can be subcritical or transcritical and have one or more pressure levels [3–11]. In most cases, the components in these cycles are assumed to be ideal or they are modeled very simplistically. Some researchers have already taken the influence of the sizing of the components into account. Madhawa Hettiarachchi et al. [12] have minimized the ratio of the total heat exchanger surface and the net power produced by the cycle. Franco and Villani [13] have optimized the cycle and the heat exchangers separately, but used an iteration to make the connection between the system level and the component level. Walraven et al. [14] have shown that it is possible to optimize the configuration of shell-and-tube heat exchangers together with the configuration of the cycle, which was extended in Walraven et al. [15], in which an air-cooled condenser was included.

☆Published version: http://dx.doi.org/10.1016/j.apenergy.2014.12.078

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A consequence of the low conversion efficiency of heat into electricity is that most of the heat, which is added to the cycle, has to be dumped into the environment. The cooling system is therefore very important in power plants powered by low-temperature heat sources. Power plants can be cooled in three ways: air cooling, water cooling with a cooling tower and direct cooling with water, of which the two first options are most often used. The auxiliary power consumption of air-cooled condensers (ACC) is about twice as high as that one for mechanical-draft wet cooling towers (WCT) used for low-temperature geothermal power plants [16]. When low condensing temperatures are used in these plants, the investment cost of a binary plant with an ACC can be 50% higher than that of a plant with a wet cooling tower for the same conversion efficiency [16]. The disadvantage of using a wet cooling tower is of course that water is consumed, which is a big drawback when water is scarce. The type of the cooling method is therefore very important in the design of a geothermal binary power plant.

The comparison between air cooling and wet cooling has already been performed in the literature. Barigozzi et al. [17] developed a model of a cogeneration power plant powered by burning waste, while the cooling system consists of both an ACC and a WCT. They found that when the environmental temperature is below 15°C, it is best to use the ACC. When the environmental temperature is higher than 15°C, both the ACC and the WCT are used. First, the ACC is used to cool down the steam and afterwards the WCT is used to cool it further down. These results are valid for high-temperature heat sources (turbine-inlet-temperature of 450°C). Mendrinos et al. [16] compared cooling methods for geothermal binary plants. They concluded that wet cooling towers are the best choice, except when water is a very scarce product or when the climatic conditions are extreme.

The above mentioned works often use simplified models of the cooling system. Other researchers have optimized the configuration of the cooling system itself. Rubio-Castro et al. [18] used the work of Kloppers and Kröger [19] to simulate and optimize the performance of a mechanical-draft wet cooling tower and compared the Merkel to the Poppe method. They repeated the optimization for different fill types. Serna-González et al. [20] performed a similar research, but defined the problem as a MINLP (Mixed Integer Non-Linear Problem) in which the type of packing and the type of draft were the integer optimization variables. They used the Merkel method to calculate the heat and mass transfer in the cooling tower.

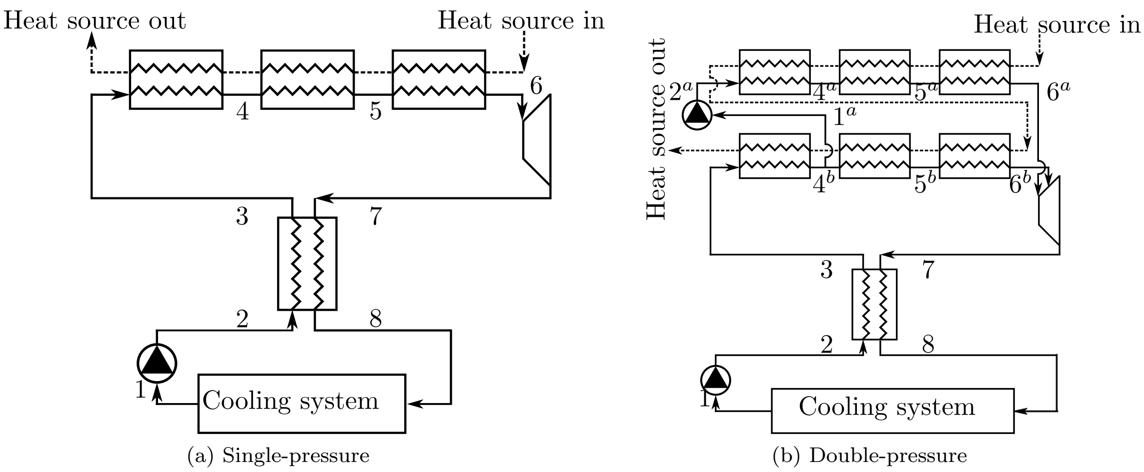
In this work we combine the three above mentioned research areas: optimization of ORCs, comparison between cooling systems and optimization of cooling systems; all at once simultaneously. In our previous work [15] we maximized the net present value of an air-cooled ORC, in which the parameters of the ORC, the configuration of the heat exchangers and the configuration of an ACC are optimized together. In this paper we add a model for a wet cooling tower based on the work of Kloppers [21] and minimize the levelized cost of electricity production (LCOE)[1] for both water-cooled and air-cooled ORCs. The results of both types of cooling are compared to each other and the influence of the brine-inlet temperature, brine-outlet temperature, discount rate and water price on the performance of the power plant are investigated.

2. Physical model

2.1. Organic Rankine cycle

Organic Rankine cycles (ORCs) can have different configurations of which a few are modeled in this paper. The cycles can be simple or recuperated, subcritical or transcritical and can have one or two pressure levels. Figures 1a and 1b give the scheme of a single-pressure, recuperated ORC and a double-pressure, recuperated ORC, respectively. In these schemes all the possible heat exchangers (economizer, evaporator, superheater and recuperator) are shown, but they are not always necessary. The cooling system can be an air-cooled condenser (ACC) (section 2.4) or a wet cooling tower (WCT) (section 2.5) connected to a condenser and if necessary a desuperheater.

[1]This is the constant electricity price needed during the lifetime of the power plant to reach brake-even at the end of the lifetime of the power plant.



(a) Single-pressure

(b) Double-pressure

Figure 1: Scheme of a single-pressure, recuperated (a) and double-pressure, simple (b) ORC.

It is assumed that state 1 is saturated liquid and that the isentropic efficiency of the pump is 80%. More information about the modeling of the cycle can be found in Walraven et al. [11].

2.2. Heat exchangers

All the heat exchangers used in this paper are of the shell-and-tube type. TEMA E type heat exchangers with a single shell pass and with the inlet and the outlet at the opposite ends of the shell are modeled. The "dirty" fluid (brine and cooling water) flows on the tube side for easy cleaning. The Bell-Delaware method [22, 23] is used to model the heat transfer and pressure drop in single-phase flow, evaporation and condensation on the shell-side. More information about the modeling of the heat exchangers can be found in Walraven et al. [14].

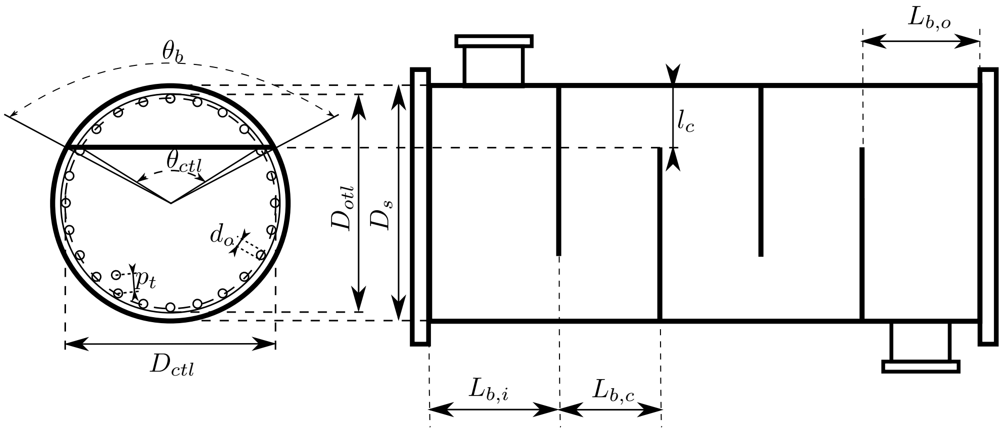


Figure 2: Shell-and-tube geometrical characteristics. Figure adapted from Shah and Sekulić [23]. See also Walraven et al. [14].

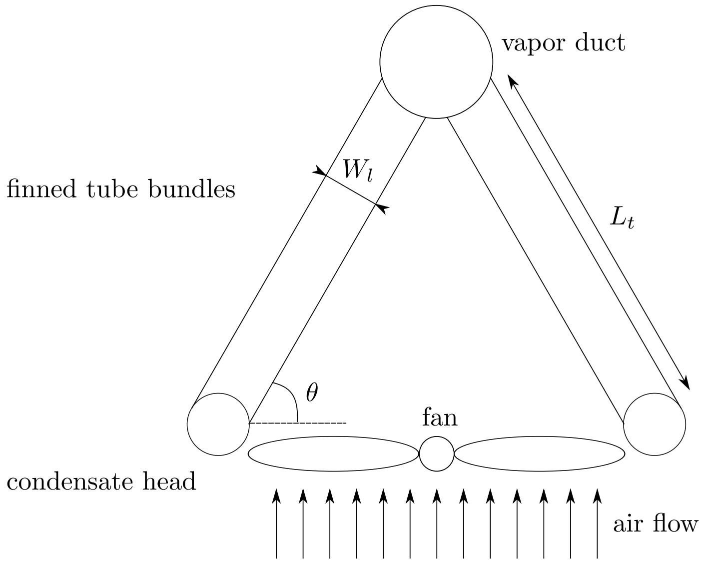
Figure 2 shows a TEMA E shell-and-tube heat exchangers with its basic geometrical characteristics. These are the shell outside diameter D_s, the outside diameter of a tube d_o, the pitch between the tubes p_t, the baffle cut length l_c and the baffle spacing at the inlet $L_{b,i}$, outlet $L_{b,o}$ and the center $L_{b,c}$.

2.3. *Turbine*

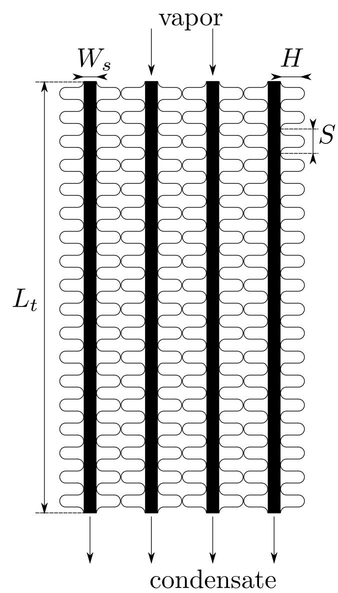
The turbine used in this paper is an axial-inflow, axial-outflow turbine. The results of Macchi and Perdichizzi [24] are used to predict the isentropic efficiency of a turbine stage as done in Walraven et al. [15].

2.4. *Air-cooled condenser*

The cooling system in figure 1 can be an air-cooled condenser (ACC). Different type of ACCs exist, but in this paper only A-frame ACCs with flat tubes and corrugated fins are used. This type is most often used in power plants because the pressure drop on the air side is lower than the one in ACCs with round tubes [25, 26]. The geometry of such an ACC and the bundle geometry of flat tubes with corrugated fins are shown in figure 3.



(a) A-frame



(b) Tube bundle

Figure 3: Geometry of an A-frame air-cooled condenser (a) and the bundle geometry of flat tubes with corrugated fins (b).

The tube-bundle geometry is determined by the tubes' small width W_s, the fin height H, the fin pitch S, the tubes' large width W_l and the length of the tubes L_t. In an A-frame ACC the tube bundles are placed at an angle θ with the horizontal. The vapor/two-phase fluid enters the condenser at the top in the vapor duct, flows down the tubes, in which it condenses, and the condensate is collected at the bottom in the condensate head. A fan at the bottom blows air over the tube bundles. The model of Yang et al. [25] is used to predict the pressure drop and heat transfer of the air-side as explained in Walraven et al. [15].

2.5. *Wet cooling tower*

Another cooling option is to use a desuperheater and a condenser, coupled to a wet cooling tower (WCT). Natural-draft cooling towers are not modeled in this paper, because they are typically used for large cooling needs. For lower cooling loads, mechanical-draft cooling towers are better suited. Both induced and forced-draft towers exist. In the former type, the fan is located downstream (at the exit of the air) of the tower, while the fan is located upstream (at the inlet of the air) in forced-draft towers. The velocity of the air at the outlet is higher for induced-draft towers and the chance of recirculation of wet air is therefore lower. This is the reason why induced-draft towers are more often used and are the focus in this work. Such an induced-mechanical-draft tower is shown in figure 4. The warm cooling water enters the tower in the sprayers in which it is sprayed over the fill. The fill is used to increase the contact surface between droplets and air in order to increase the heat and mass transfer. At the bottom the cooled water is caught and it is sent back to cool the ORC condenser. The air flows in the other direction; it enters the tower from the sides at the bottom and flows through the inlet louvers. These louvers are used to prevent the inflow of unwanted elements, to prevent water splash and to decrease the amount of sunlight irradiation. When flowing upwards in the tower, the air heats up and the humidity increases. The drift eliminators are used to decrease the number of water droplets taken by the airflow. The height of the inlet H_i, the height of the fill H_{fi}, the height of the spray zone H_{sp} and the width of the tower W_t are shown in the figure.

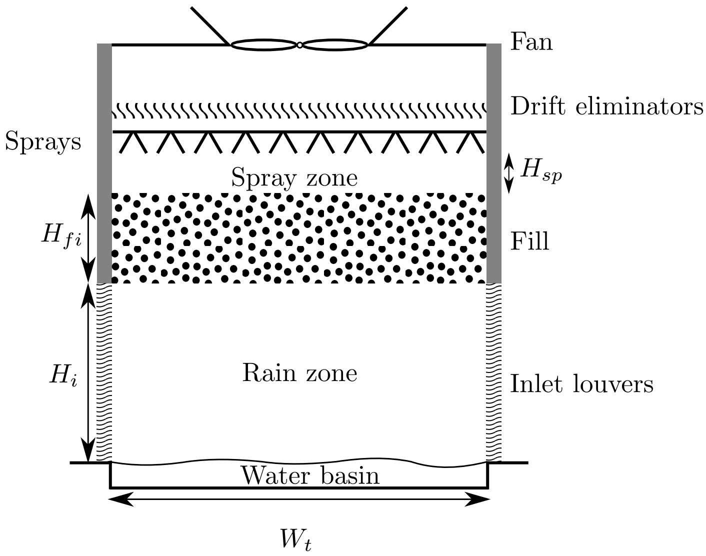


Figure 4: Geometry of an induced mechanical-draft wet cooling tower. Figure adapted from Kloppers [21].

In this paper, only square towers with a film fill packing are modeled. The model of the cooling tower is based on the work of Kloppers [21]. He developed empirical correlations for the performance of different fill types and used a fourth order Runge-Kutta method to solve the Poppe equations. The pressure drop of the air in the tower is also based on the above mentioned work.

For a given inlet temperature and required outlet temperature of the cooling water, the required Merkel number[2] is calculated. Based on the geometry of the tower, the properties of the air and the water, the Merkel number in the spray zone and the rain zone can be calculated. From this the required Merkel number in the fill follows and the height of the fill can be calculated. To calculate the electrical power consumption of the fan, the pressure drop of the air in the inlet zone, the inlet louvers, rain zone, fill support structure, fill, spray zone, water distribution, drift eliminator and fan upstream are calculated [21]. It is assumed that the fan has an efficiency of 60%.

[2]The Merkel number is a non-dimensional parameter describing the transfer characteristics in the cooling tower, defined as $Me = \frac{h_d a_{pa} L_{pa}}{G_w}$ with h_d the mass transfer coefficient, a_{pa} the area density of the packing and G_w the mass velocity of the water.

3. Economics

3.1. Levelized cost of electricity

The levelized cost of electricity (LCOE) is the constant electricity price needed during the lifetime of the power plant to reach break even over the lifetime of the project. The LCOE is calculated in €/MWh as [27]

$$\text{LCOE} = \frac{C_{EPC} + \sum_{t=1}^{t_{LT}} \left[\left(C_{O\&M,t} + C_{water,t}\right) (1+i)^{-t} \right]}{\sum_{t=1}^{t_{LT}} \dot{W}_{net}\ N\ (1+i)^{-t}}, \tag{1}$$

with C_{EPC} the engineering, procurement & construction overnight cost (EPC) of the installation, t_{LT} the lifetime of the installation, $C_{O\&M,t}$ the operations and maintenance cost in year t which is assumed to be 2.5% of the investment cost of the ORC per year [2], $C_{water,t}$ the water cost in year t, $\dot{W}_{net}$ the net electric power output, which takes an electric generator efficiency of 98% into account, expressed in MW_e, N the number of full-load hours per year (an availability of 95% is assumed) and i the discount rate. The EPC cost consists of two parts: the cost of the drilling $C_{drilling}$ and the cost of the ORC C_{ORC} (see section 3.2).

3.2. Cost of ORC

The overnight EPC investment cost of the ORC, C_{ORC}, can be calculated as:

$$C_{ORC} = \sum_i \left(f_{M,i} f_{P,i} f_{T,i} + f_I\right) C_{E,i}, \tag{2}$$

with $C_{E,i}$ the equipment cost of component i and $f_{M,i}$, $f_{P,i}$ and $f_{T,i}$ correction factors (all ≥ 1) for non-standard material, pressure and temperatures, respectively. f_I is an average installation-cost factor [28]. This installation-cost factor includes the costs for erection, instrumentation and control of the power plant and is about 0.6 [28, 29]. Correlations for the equipment cost $C_{E,i}$ are given in table 1.

Component	Capacity measure	Size range	Cost correlation	Ref
Shell-and-tube heat exchanger	A [m^2]	80-4000 m^2	$3.50\ 10^4 \left(\frac{A}{80}\right)^{0.68}$ [€^{2013}]	[28]
Centrifugal pump (incl. motor)	$\dot{W}_{pump}$ [kW]	4-700 kW	$10.51\ 10^3 \left(\frac{\dot{W}_{pump}}{4}\right)^{0.55}$ [€^{2013}]	[28]
Air-cooled heat exchanger	Bare-tube A [m^2]	200-2000 m^2	$1.67\ 10^5 \left(\frac{A}{200}\right)^{0.89}$ [€^{2013}]	[28]
Fan (incl. motor)	$\dot{W}_{fan}$ [kW]	50-200 kW	$1.31\ 10^4 \left(\frac{\dot{W}_{fan}}{50}\right)^{0.76}$ [€^{2013}]	[28]
Turbine	$\dot{W}_{turbine}$ [kW]	0.1-20.0 MW	$-1.66\ 10^4 + 716\dot{W}_{turbine}^{0.8}$ [€^{2013}]	[30]
Film fill packing	V_{pack} [m^3]	/	$41.57\ V_{pack}$ [€^{2013}]	[21]
Structure tower	Outside A [m^2]	/	$332.56\ A$ [€^{2013}]	[21]

Table 1: Cost correlation for the different components. The data from Smith [28], Towler and Sinnott [30] and Kloppers [21] are adapted taking into account that 1€=1.35$ and with a Chemical Engineering (CE)-index of 564 in July 2013. The reference CE-index of 100 was set in the base period 1957-1959. CE-indices can be found on `http://www.che.com/pci/`.

The cost correlations in table 1 are valid for carbon steel, for design temperatures between 0 and 100°C and for design pressures between 0.5 and 7 bar. Such "normal" designs are good enough for most components in a low-temperature ORC. Only the heat exchangers between brine and working fluid operate at higher pressures and temperatures and have a higher risk for corrosion. For these heat exchangers, the values of table 1 are adjusted using the above mentioned correction factors; the tubes are made from stainless steel ($f_M = 1.7$), work at higher pressures ($f_P = 1.5$) and at higher temperatures ($f_T = 1.6$) [28].

The economic computation here considers $C_{drilling}$ as a given and our computations do not consider financing costs (interest during construction), nor owners costs and provisions for contingencies. The computed LCOE is kind of a "naked" LCOE based on the EPC overnight cost.

4. Optimization

The objective of the optimization is to minimize the levelized cost of electricity (LCOE), by finding the optimal conversion cycle parameters and sizing. This optimization is performed with the use of the CasADi [31] and WORHP [32] software. The models themselves are developed in Python and the fluid properties are obtained from REFPROP [33]. We perform a *system* optimization which means that the cycle parameters and the configuration of all the components are optimized together.

4.1. Optimization variables and constraints

The optimization variables of a single-pressure, recuperated cycle are the temperature before the turbine, the saturation temperature at the pressure before the turbine, the pressure at the inlet of the pump, the mass flow of the working fluid and the effectiveness of the recuperator. For double-pressure cycles, the temperature before the second turbine, the saturation temperature at the pressure before the second turbine and the mass flow rate through the second turbine are added. More information about these optimization variables can be found in Walraven et al. [14].

The optimization variables of each shell-and-tube heat exchanger are the shell diameter D_s, tube-outside diameter d_o, tube pitch p_t, baffle cut l_c and the distance between the baffles $L_{b,c}$ [14].

The fin height H, the fin pitch S, the air velocity at the minimum cross section V_{Amin} and the number of tubes n_{tubes} are the optimization variables of the ACC and a non-linear constraint is used to limit the length of the tubes, as done in Walraven et al. [15].

Table 2 shows the optimization variables used for the wet cooling tower and their lower and upper bounds. T_{wb} is the wet-bulb temperature. The height of the spray zone H_{sp} is fixed at 0.5 m. The height of the fill H_{fi} is a result of the cooling-tower model and is therefore not an optimization variable.

Optimization variable	Lower boundary	Upper boundary
Tower width W_t	1 m	40 m
Inlet height H_i	1 m	20 m
Relative air mass flow $\dot{m}_{air}/\dot{m}_{brine}$	1.5	500
Relative cooling fluid mass flow $\dot{m}_{cf}/\dot{m}_{brine}$	1.5	500
Minimum cooling-fluid temperature T_{cf}^{min}	T_{wb}	/

Table 2: Optimization variables used for the wet cooling tower and their lower and upper boundaries.

5. Results

5.1. Parameters of the reference case

The parameters of our "reference" case are given in table 3, which are based on a proposed geothermal demonstration project in Belgium. In the next subsections, the influence of many of these parameters (brine-inlet temperature, brine-outlet temperature, number of pressure levels, water price, yearly water-price evolution and climate conditions) on the performance of the ORC is investigated. For each of the parameter variations, a new design optimization is performed with the optimization variables described in section 4.1 to obtain the minimum LCOE.
Many of the above mentioned economic parameters used in this paper are based on the literature and the economic analysis is therefore not detailed enough to be used for a business plan. We only focus on electricity production, while using the geothermal source for heating purposes can improve the economics of the project.

Well parameters	
Brine wellhead temperature	125°C
Brine production	194 kg/s
Well pumps consumption	600 kW_e
Wells cost	27.5 M€

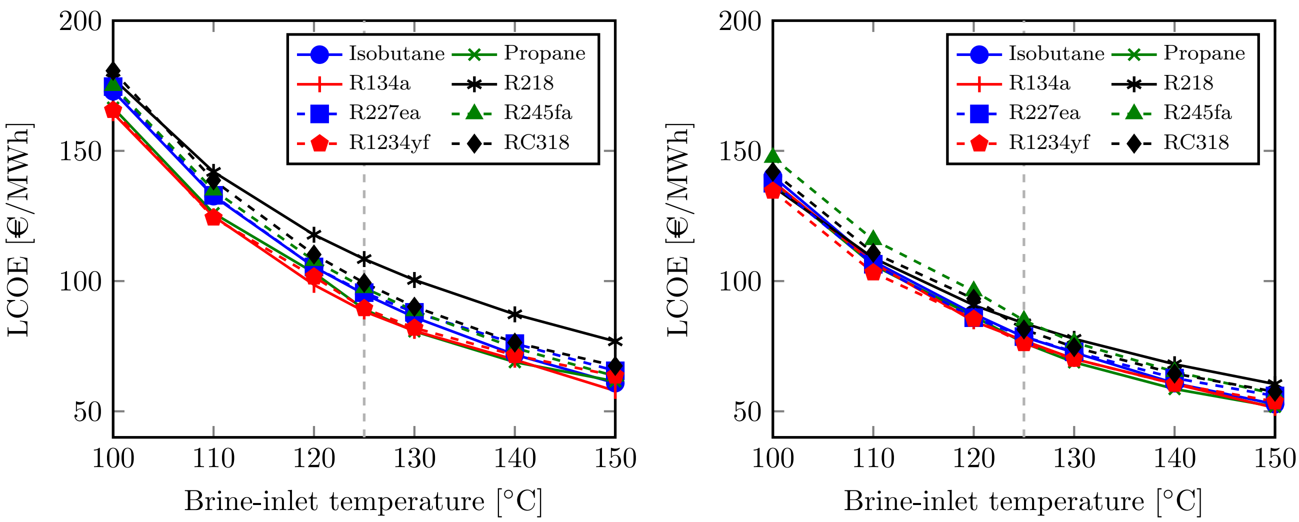
Economic parameters	
Lifetime plant	30 years
Discount rate	4%/year
Water price	0.5€/m^3

Environmental conditions	
Dry-bulb temperature	10.3°C
Wet-bulb temperature	8.6°C
Air pressure	1016hPa

Table 3: Parameters of the reference case

5.2. Influence of the brine-inlet temperature

The influence of the brine-inlet temperature on the levelized cost of electricity production for ORCs with air cooling and water cooling are shown in figures 5a and 5b, respectively.



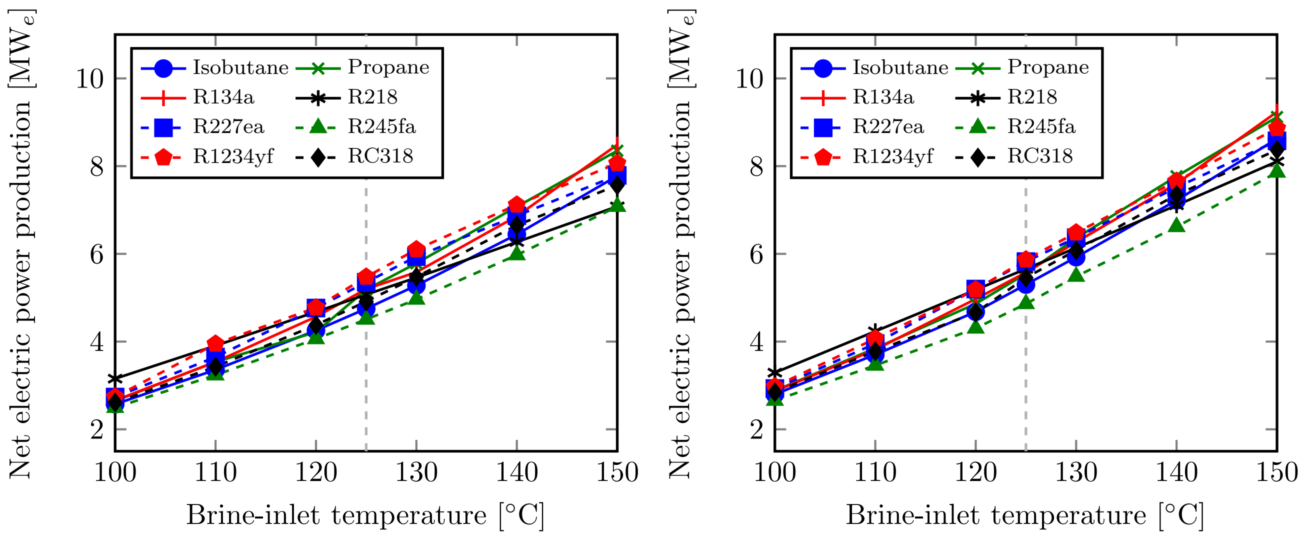
(a) Single-pressure recuperated cycle with ACC

(b) Single-pressure recuperated cycle with WCT

Figure 5: LCOE of a single-pressure recuperated ORC with an ACC (a) and a WCT (b) powered by geothermal heat, for various thermodynamic-cycle fluids and for varying inlet temperatures of the brine.

The LCOE decreases with increasing brine-inlet temperature as expected, influenced by the net electric power produced by the installation (figure 6) and thus the investment cost of the installation (specific cost of the ORCs are shown in figure 7). Clearly, the net electric power production increases strongly with the brine-inlet temperature because of the increased plant efficiency, which results in a higher gross turbine power and lower cooling needs.

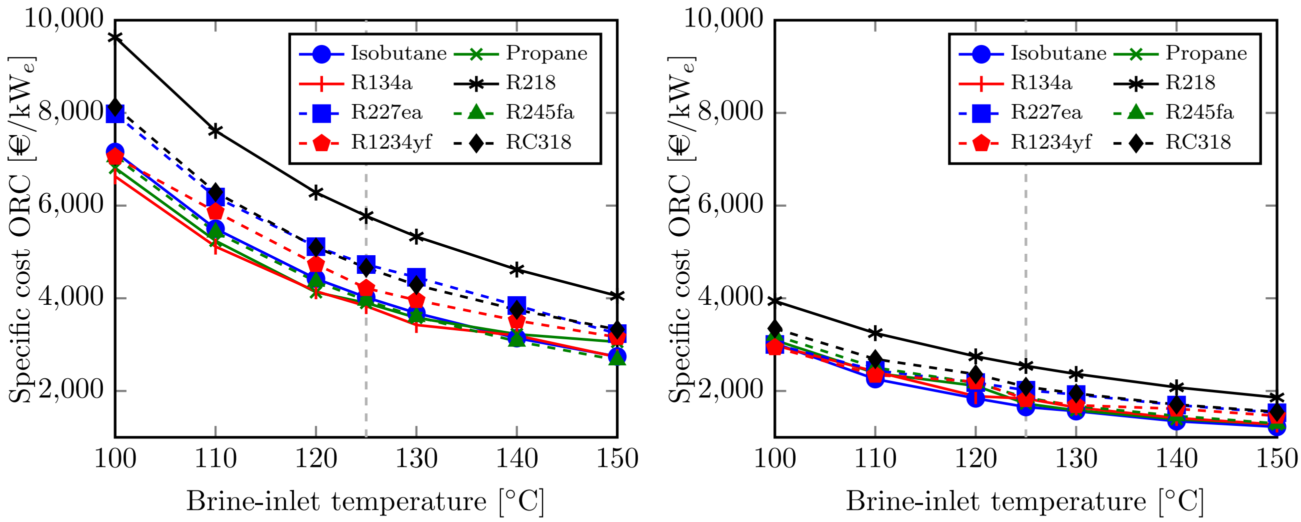
The comparison of figures 5a and 5b shows clearly that the LCOE for ORCs with a WCT is lower than the one for ORCs with an ACC, and especially for lower brine-inlet temperatures. The net electric power production of ORCs with a WCT (figure 6b) is (slightly) higher than the one with an ACC (figure 6a) because of the lower condenser temperature and the lower auxiliary power consumption, but it is especially the lower (specific) investment cost which results in a better LCOE for the ORCs with a WCT. Indeed, the comparison of figures 7a and 7b shows that the specific cost of ORCs with a WCT is often less than 50% of the specific cost of ORCs with an ACC. This is caused by the high investment cost of an ACC.



(a) Single-pressure recuperated cycle with ACC

(b) Single-pressure recuperated cycle with WCT

Figure 6: Net electric power production of a single-pressure recuperated ORC with an ACC (a) and a WCT (b) powered by geothermal heat, for various thermodynamic-cycle fluids and for varying inlet temperature of the brine.



(a) Single-pressure recuperated cycle with ACC

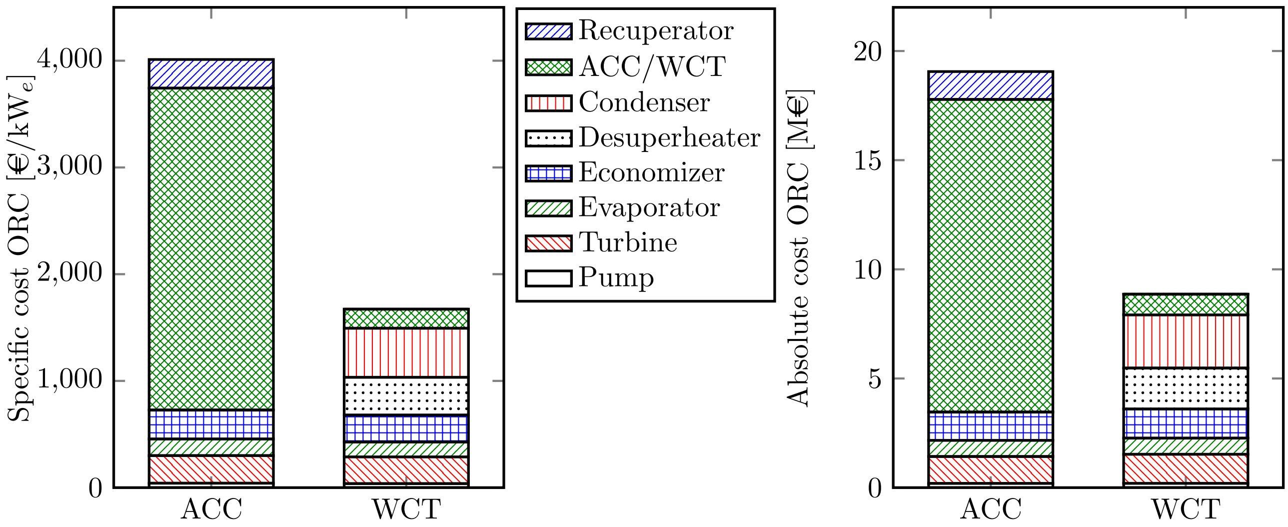
(b) Single-pressure recuperated cycle with WCT

Figure 7: Specific cost of a single-pressure recuperated ORC with an ACC (a) and a WCT (b) powered by geothermal heat, for various thermodynamic-cycle fluids and for varying inlet temperature of the brine.

Figures 8a and 8b show the distribution of the specific and absolute cost of an ORC with isobutane and a brine-inlet temperature of 125°C. Two big differences catch the eye. The first one is the very high cost of the ACC in comparison with the WCT. The cost of the ACC accounts for about 80% of the total cost of the ORC, while the cost of the WCT is about one third of the total cost if that option is chosen. The cost of the ORC accounts for 41% and 24% of the total project cost (including the costs of the wells) for air cooling and water cooling, respectively. The second notable result is the absence of a recuperator in the

ORC with a WCT. The installation of a recuperator can decrease the cooling needs, which results in a lower investment and operational cost of the cooling installation as explained in Walraven et al. [15]. The extra cost of the recuperator is compensated for by the lower cost of the cooling system when an ACC is used, but not when a WCT is used.

The turbine costs about 1.2 and 1.3 M€ for the ORC with an ACC and a WCT, respectively. Another expensive component, apart from the cooling system, is the economizer, which costs about 1.3M€ for both cases. The evaporator is relatively inexpensive (0.7 M€) because of the high heat transfer coefficient during evaporation (see for example Walraven et al. [14]) and the high average temperature difference in the evaporator.



(a) Specific cost

(b) Absolute cost

Figure 8: Distribution of the specific cost (a) and the absolute cost (b) for single-pressure, simple and recuperated ORCs with isobutane as the working fluid for a brine-inlet temperature of 125°C. The cost of the superheater is very small in all cases and is not shown in the figure.

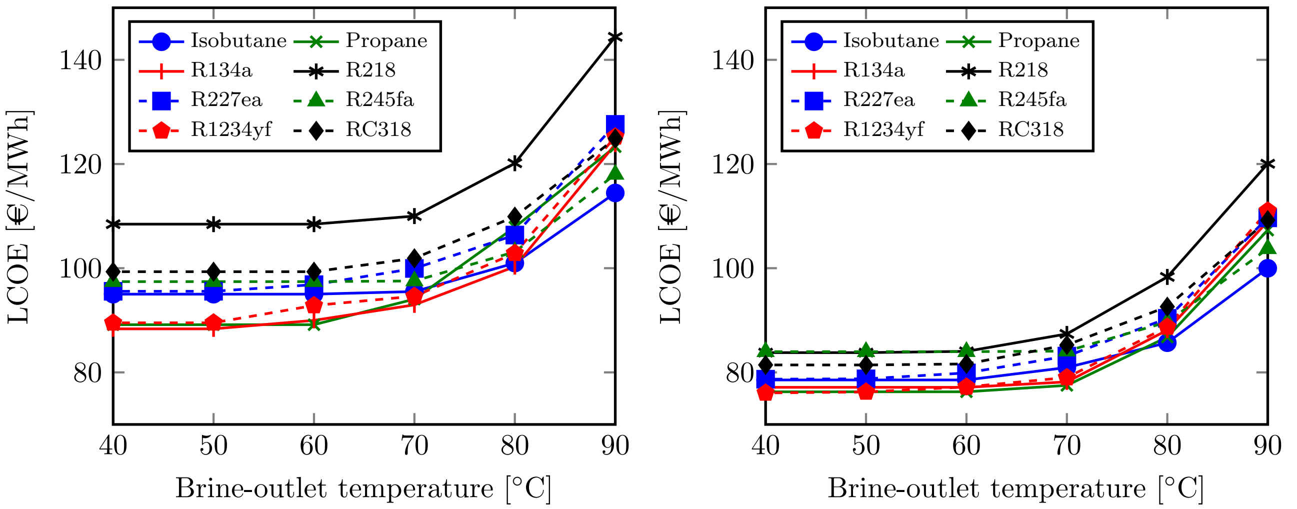
5.3. Influence of the brine-outlet temperature

In this section the effect of a constraint on the brine-outlet temperature is investigated, although only electricity production from the geothermal source is taken into account and direct use of the geothermal source for heating purposes is not considered in this paper.

Figures 9a and 9b show the influence of the brine-outlet-temperature constraint on the LCOE for ORCs with an ACC and a WCT, respectively. The LCOE remains constant for both cooling methods up to a brine-outlet temperature of about 60-70°C, which is the optimal brine-outlet temperature when no constraint is used. For higher values of the constraint, the LCOE starts to increase and it becomes interesting to use a recuperator in all cases. Such a recuperator can increase the cycle efficiency and when the heat input to the cycle is limited (constraint on brine-outlet temperature), the net power output can also increase [11]. This last effect is more important than the increase in cost due to the extra heat exchanger.

5.4. Impact of 2 pressure levels

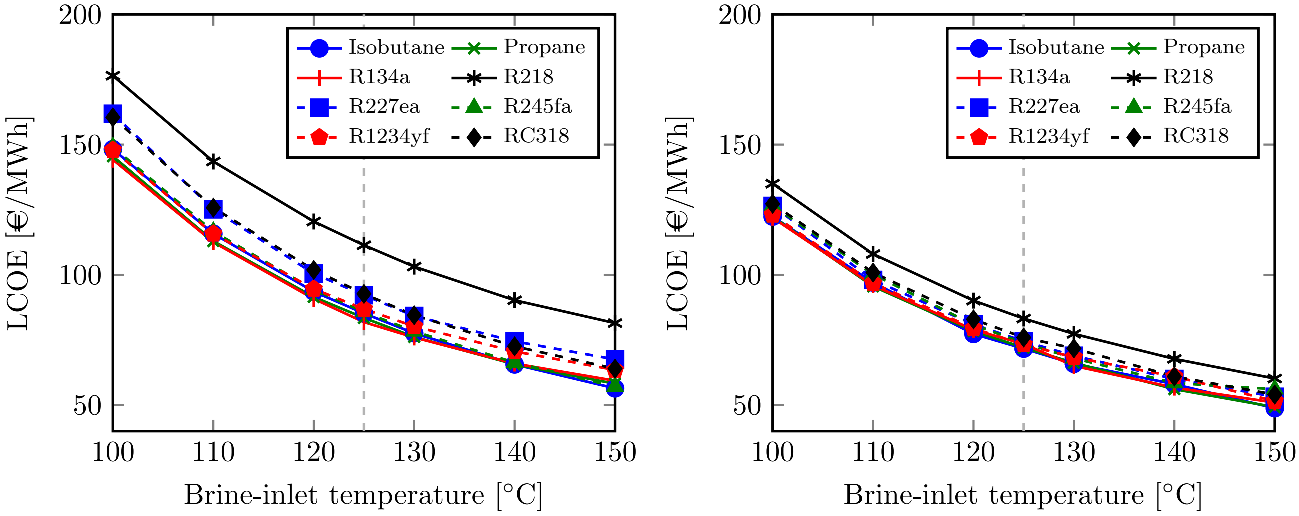
One way to improve the efficiency of a single-pressure ORC is by using more than one pressure level [11]. In this section double-pressure ORCs are compared to single-pressure ones and the influence of the brine-inlet temperature is investigated.



(a) Single-pressure recuperated cycle with ACC

(b) Single-pressure recuperated cycle with WCT

Figure 9: LCOE of a single-pressure recuperated ORC with an ACC (a) and a WTC (b) powered by geothermal heat, for various thermodynamic-cycle fluids and for varying outlet temperatures of the brine.



(a) Double-pressure recuperated cycle with ACC

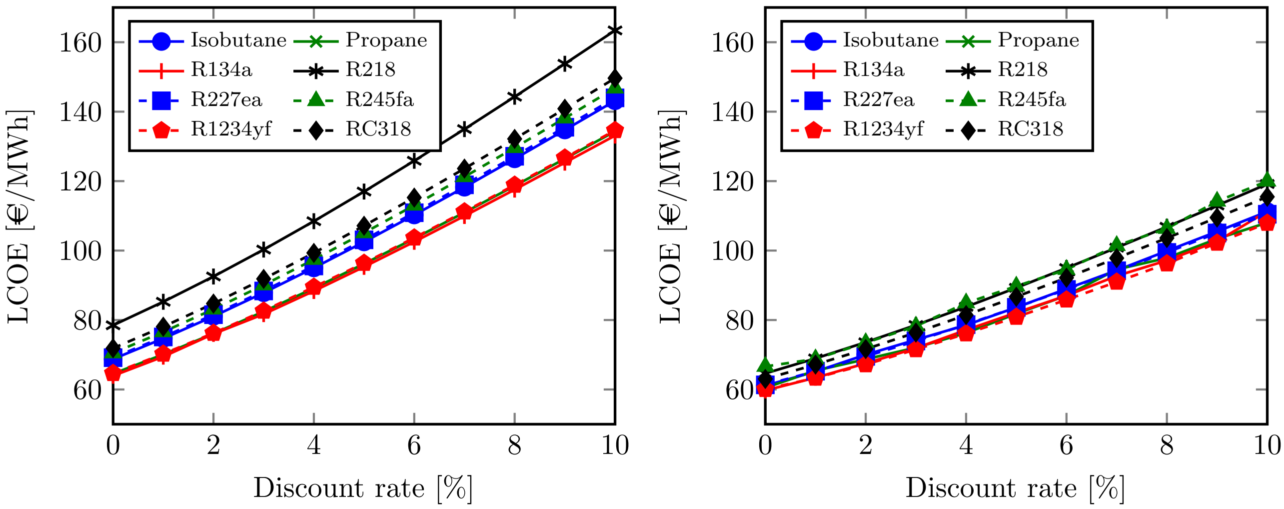
(b) Double-pressure recuperated cycle with WCT

Figure 10: LCOE of a double-pressure recuperated ORC with an ACC (a) and a WCT (b) powered by geothermal heat, for various thermodynamic-cycle fluids and for varying inlet temperatures of the brine.

Figures 10a and 10b show the influence of brine-inlet temperature on the LCOE of double-pressure ORCs with an ACC and a WCT, respectively. Comparison with figure 5 shows that the LCOE of double-pressure cycles is considerably lower than the one of single-pressure cycles, because the net electric power output of the ORC can increase by adding another pressure level [11]. This effect is more important than the additional cost of the extra pressure level.

5.5. Influence of the discount rate

A geothermal power plant requires a high investment in the beginning of the project and the revenues are obtained in the (far) future. The discount rate is therefore an important economic factor. In this section the discount rate is varied between 0 and 10%, while using for all other parameters the reference value (table 3).



(a) Single-pressure recuperated cycle with ACC

(b) Single-pressure recuperated cycle with WCT

Figure 11: LCOE of a single-pressure recuperated ORC with an ACC (a) and a WTC (b) powered by geothermal heat, for various thermodynamic-cycle fluids and for varying discount rates.

As seen from figure 11, the value of the discount rate has a very strong influence on the LCOE, especially for ORCs cooled by an ACC. ORCs with an ACC require a higher investment cost than ORCs with a WCT (figure 7) and the effect of the discount rate is therefore more important if an ACC is used. For low values of the discount rate, the LCOE of an ORC with an ACC is only slightly higher than the one of an ORC with a WCT, while the difference is much larger for high values of the discount rate.

The results show that the optimal configuration of the power plant – and therefore also the net electric power output, investment cost, etc. – is almost independent of the discount rate. This is again due to the high investment cost and the low operational cost. As seen from equation (1), minimizing the LCOE is almost equal to minimizing the fraction of the investment cost to the net electric power production when the operational cost is low. Consequently, the discount rate influences only the value of the LCOE, but has almost no influence on the optimal configuration of the power plant.

5.6. Influence of the water price and yearly water-price evolution

A WCT consumes water and the water price and the yearly water-price evolution have therefore an influence on the power plant. These parameters have no effect on the performance of an ACC and only the results of ORCs with a WCT are therefore given in this section.
Figures 12a and 12b show the influence of the water price and the yearly water-price evolution on the LCOE of the power plant. The LCOE increases as expected if water becomes more expensive (now and/or in the future). Comparing figure 12a to figure 5a shows that ORCs with an ACC can become more interesting than ORCs with a WCT when water is expensive; in this reference case, if water were to cost more than 1€/m^3.

If water becomes more expensive, cooling in general costs more. To counteract this, the optimizer tries

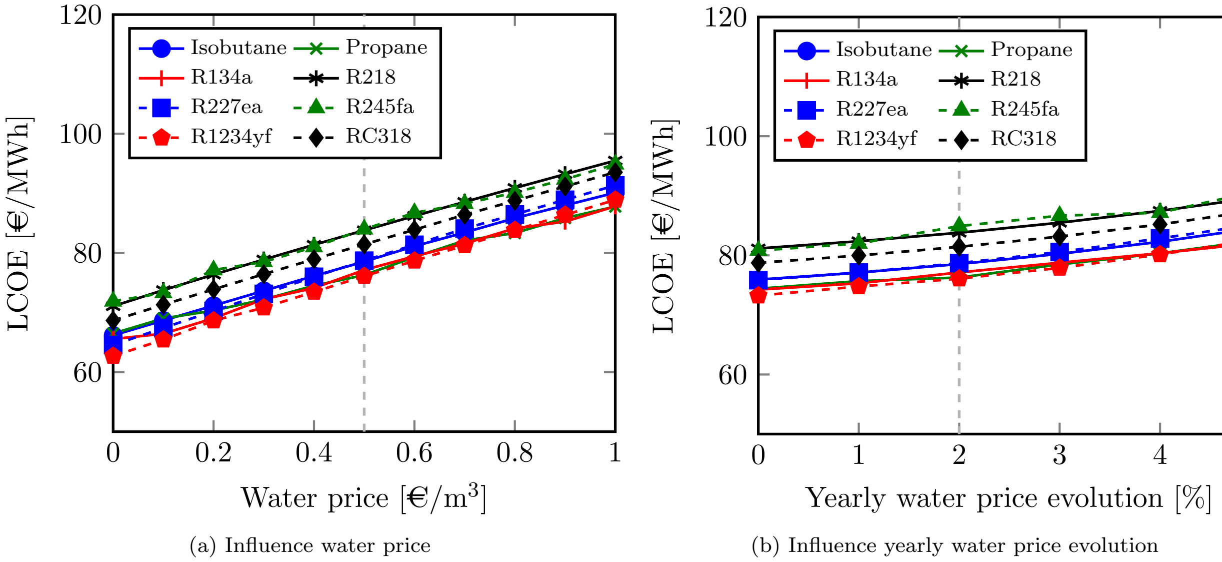


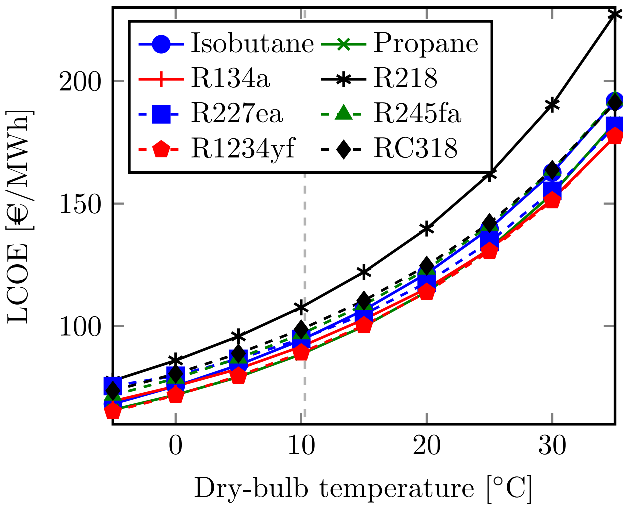
Figure 12: LCOE of a single-pressure recuperated ORC with a WTC powered by geothermal heat, for various thermodynamic-cycle fluids and for varying water price (a) and yearly water price evolution (b). The dashed vertical line represents the reference value of the water price.

to reduce the water consumption by reducing the cooling needs and by consuming less water in the cooling tower. The first effect is obtained by increasing the cycle efficiency and by increasing the brine-outlet temperature. The drawbacks are that the plant efficiency, and therefore also the net power output, decreases slightly and that the investment cost increases. To reduce the water consumption in the cooling tower, the mass flow rate of cooling water is reduced. To compensate for the negative effect this has on the performance of the ORC [34], the minimum cooling-water temperature is reduced. In order to obtain this, the height of the packing is increased, resulting in a higher investment cost and a higher pressure drop (auxiliary power consumption).

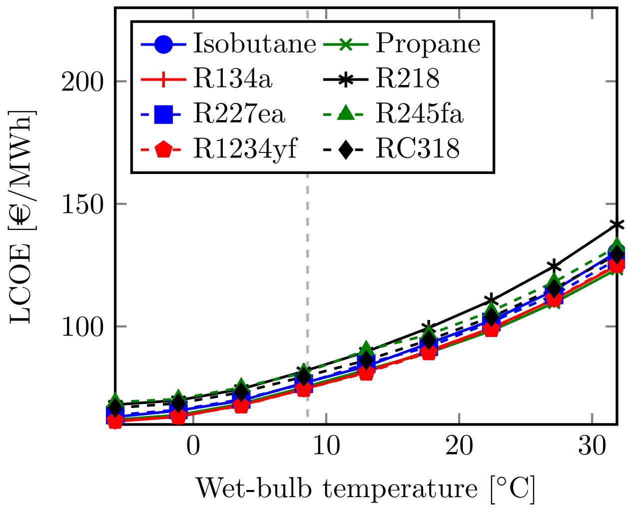
5.7. Influence of the climate conditions

In this section, the influence of the climate conditions on the performance of the power plant is investigated. The dry-bulb temperature is varied between 0 and 35°C, while keeping the relative humidity constant at 80% as in the reference case. All other parameters are the reference ones, as given in table 3. The effect of the dry-bulb temperature on the LCOE of the investigated ORCs is shown in figure 13. For the wet-cooled ORCs, the LCOE is given as a function of the wet-bulb temperature, because this temperature is most determining for wet cooling.

Comparison of figures 13b and 13b shows that the LCOE is always lower for water-cooled ORCs than for air-cooled ORCs for the investigated climate conditions. Especially for high dry-bulb temperatures, the ORCs with a WCT perform much better than ORCs with an ACC. This is again due to the very high investment cost for ACCs. For low dry-bulb temperatures, the difference between air cooling and wet cooling is much lower because of two reasons. First, the lower the dry-bulb temperature, the lower the condensation temperature and the higher the cycle efficiency. So, less cooling is needed. Second, the cooling-water temperature in the wet-cooled ORCs is limited from below to avoid freezing. In this paper, the minimum allowed cooling-fluid temperature is 5°C. So, the condensation temperature of ORCs with a WCT has a lower limit, while this is not the case for ORCs with an ACC.



(a) Single-pressure recuperated cycle with ACC



(b) Single-pressure recuperated cycle with WCT

Figure 13: LCOE of a single-pressure recuperated ORC with an ACC (a) or a WCT (b) powered by geothermal heat, for various thermodynamic-cycle fluids and for varying dry-bulb temperature and a constant relative humidity (80%). The results for the ORCs with a WCT are given as a function of the wet-bulb temperature. The dashed vertical line represents the reference value of the water price.

6. Conclusions

In this paper, a system optimization of both air-cooled and water-cooled ORCs power by geothermal heat is performed. In order to achieve the minimum levelized cost of energy (LCOE), the cycle parameters of the ORC, the geometry of the heat exchangers and the geometry of the cooling system are optimized together.

Comparison of water cooling to air cooling in ORCs shows that the former type results in better economics, because of the increased net electric power output and especially because of the much lower investment cost. This higher investment cost also results in a much higher impact of an increasing discount rate on the LCOE.

The difference between the two types of cooling decreases with increasing brine-inlet temperature. This is because the efficiency of the ORC increases and the cooling needs decrease (relatively speaking). Air cooling can become interesting if water is very expensive (more than 1€/m^3 in our reference case).

The brine-inlet temperature and the dry-bulb temperature of the surroundings are the most influencing parameters. If the brine-inlet temperature increases from 100 to 150°C, The LCOE decreases from about 170 to about 60 €/MWh for air-cooled ORCs and from about 140 tot about 55 €/MWh for water-cooled ORCs for the investigated reference parameters. For the dry-bulb temperature increasing from -5 to 35°C, the LCOE increases from about 70 to about 185 €/MWh and from about 65 to about 125 €/MWh for air-cooled and water-cooled ORCs, respectively.

It is also shown that the addition of an extra pressure level can improve the economics and that a constraint on the brine-outlet temperature can have a negative influence on the economics of the electric power plant. For air-cooled ORCs with dry fluids a recuperator is always useful. For the other types of cycles, a recuperator only becomes interesting when the constraint on the brine-outlet temperature is high enough.

Nomenclature

Greek Roman

η	Efficiency [-]
θ	Tube bundle angle [°]

A	Surface area [m^2]
C	Cost [€]
d_o	Tube outside diameter [m]
D_s	Shell diameter [m]
f	Correction factor [-]
H	Fin height [m]
H_x	Height of x [m]
i	Discount rate [%]
I	Income [€]
l_c	Baffle cut length [m]
L_b	Baffle spacing [m]
LCOE	Levelized cost of electricity [€/MWh]
L_t	Length of the tubes (ACC) [m]
$\dot{m}$	Mass flow [kg/s]
MINLP	Mixed integer non-linear problem
N	Number of full load hours [-]
n_{tubes}	Number of tubes [-]
ORC	Organic Rankine cycle
p	Price [€]
p_t	Tube pitch [m]
S	Fin pitch [m]
T	Temperature [°C]
t	Time [year]
V_{Amin}	Velocity at minimum flow area [m/s]
$\dot{W}$	Mechanical power [kW]
W_s	Tube small width [m]
W_t	Tower width [m]
W_l	Tube large width [m]

Sub-and superscripts

air	Air
$brine$	Brine
$drilling$	Drilling
E	Equipment
el	Electrical
EPC	Engineering, Procurement and Construction
fan	Fan
I	Installation
in	Inlet
LT	lifetime
M	Material
net	Nett
OM	Operation and maintenance
ORC	ORC
P	Pressure
$pump$	Pump
T	Temperature
$turbine$	Turbine

Acknowledgments

Daniël Walraven is supported by a VITO doctoral grant.

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